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FIRST LAW ANALYSIS OF A COMPRESSOR USING A COMPUTER SIMULATION MODEL

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ABSTRACT

To perform a second law analysis for a compressor, basic thermodynamic variables must be known. This data set can be generated by an analysis requiring use of the first law. This aspect is the focus of the paper. In developing a computer simulation model considerable attention has been given to the proper evaluation of cylinder thermodynamics and cyclic heat transfer. This model compares well with the polytropic process model and with a benchmark simulation except for the cylinder temperature-time history. This is because the instantaneous temperature prediction strongly depends on mass flow and heat transfer rates.

INTRODUCTION

For many industrially important machines, including reciprocating compressors which are the focus of the present paper, design modifications leading to more efficient operation can be inferred by consideration of a rank-ordered list of availability (exergy) losses and destructions experienced over typical operating cycles. The development of such a list is the result of a second law analysis.

Second law analyses of machines modeled as control volumes reported in the literature are generally for operation at steady-state. A steady-state analysis is not appropriate for reciprocating compressors and this introduces some computational and theoretical complexities due to the often strongly time dependent heat transfer and mass flow rates. Moreover, to perform second law analyses requires that thermodynamic variables such as pressure, temperature, enthalpy and entropy be known at various points of the operating cycle. Accordingly, the objective of the current phase of the research project being reported upon here is not on the ultimate goal of second law analysis but rather on the intermediate, but essential, step of developing adequate means for accurately predicting, via conservation of mass, momentum and energy, the needed operating variables. This paper presents a progress report of the current phase of the project, including typical results and a brief outline of future directions.

SIMULATION MODELS

First Law Analysis

For thermodynamic analysis of the cylinder we choose the control volume as shown in Figure 1. We assume the following: (i) uniform gas properties throughout the cylinder, (ii) ideal gas (pv = RT) with constant specific heats, (iii) no leakage past the piston, (iv) uniform running speed ($\dot{\theta}$ = constant), (v) negligible oil and friction effects, (vi) negligible kinetic and potential energies associated with mass flux, and (vii) no heat exchange between the cylinder and suction (or discharge) gas.

Using ideal gas property relations, the first law of thermodynamics yields the following equation:

$$m_c c_v \dot{T}_c + c_v \dot{m}_c T_c = \dot{Q} - p_c \dot{V}_c + \dot{m}_s c_p T_s - \dot{m}_d c_p T_d \quad (1)$$

where
$$\dot{m}_c = \dot{m}_s - \dot{m}_d \quad (2)$$

$$\dot{Q}(t) = U(t)A(t)[T_a - T_c(t)] \quad (3)$$

In order to compute U(t), the instantaneous cylinder heat transfer coefficient $h_c(t)$ is required. A number of formulations mostly based on engines [1,6,7,12-18] are available; also, a formulation by Adair et al [7] is available which was developed from the compressor experimental data.

Polytropic Process Model

For real or ideal gases, the compression or expansion process can often be described by

$$p_c v_c^n = p_o v_o^n \quad (4)$$

where n is an empirical polytropic index. In the specific case of an ideal gas, use of the ideal gas equation of state leads to the following expression for cylinder temperature T_c,

$$T_c = T_o (p_c/p_o)^{(n-1)/n} \quad (5)$$

The polytropic model has been found to be an attractive substitute for the detailed first law analysis by a number of investigators [5,10,19,20]. It is easy to use and predicts pressure-time history well; however there is considerable doubt over its suitability for instantaneous temperature prediction. For example, Lee and Smith [4] show that the cyclic heat transfer between the compressor wall and the gas must be considered as the entropy generation associated with it is substantial.

Single Cylinder Simulation

A single cylinder simulation model has been developed to conduct thermodynamic analysis using either the first law or polytropic model, Eq. (4). This simulation consists of the following process/component models [19,20] and is executed using the IBM-CSMP III Simulation Language.

a. Cylinder kinematics: For $V_c(\theta)$, where $\theta = \omega t$

b. Suction reservoir at p_s and T_s

c. Discharge reservoir at p_d and T_d

d. Fluid flow through valves: For $\dot{m}_s$ and $\dot{m}_d$

e. Valve dynamics: For q_s and q_d; using single degree of freedom formulation.

Validation

In order to establish the validity of our simulation model, we compare our predictions as shown in Figure 2 with Reference [20]. This benchmark simulation model differs from our single cylinder model in the following manner: a two-cylinder model with manifold pulsation description; polytropic process for cylinder with real equation of state (R-22); heat gain by suction from cylinder, motor, and discharge; motor torque-speed-efficiency model; and, a model for leakage past the piston. Thus the benchmark model is very detailed except for the cylinder process. From Figures 2a and 2b we note the excellent correlations between the benchmark simulation and our simulation using first law/polytropic model for p_c and m_c. But, discrepancies are evident in Figure 2c for T_c. In this figure we are also presenting our polytropic model with $z = 0.9$, where $z = pv/RT$; this comparison illustrates the discrepancy between our polytropic model and the benchmark polytropic model. However, these time histories are different from the one predicted by the first law.

SOME PARAMETRIC STUDIES

To examine the differences between the first law and polytropic models, some parametric studies have been conducted.

Cylinder Heat Transfer Coefficient $h_c(\theta)$

Figure 3a compares a number of available correlations for cyclic heat transfer coefficients. We note that the h_c profiles for most of the empirical correlations [12-17], with the exception of Adair et al [7], are somewhat similar but with different magnitudes. Figure 3b compares $\dot{Q}$ using these h_c values; and these curves are virtually identical, except for the Adair et al correlation [7]. The overall heat transfer, as shown in Figure 3b, is so low that it could be ignored. But this observation is not compatible with some of the limited data available in the literature [1,4]; for example, Lee and Smith [4] have found the cyclic heat flux to be quite high. Therefore the available formulations do not adequately describe the heat transfer. More fundamental work needs to be done in this area. In particular, the radial temperature gradients reported in [4] should be included in any further analysis.

First Law vs. Polytropic Model

Figure 4 shows comparisons between the first law analysis and polytropic model with different n values (1.1 - 1.6) for air. It should be noted that the suction/discharge valve parameters have been judiciously chosen here such that the T_c profile predicted by the first law looks similar to those predicted by the polytropic models. Figure 4 shows that the p_c and T_c time histories for $n = 1.1$ and $n = 1.6$ envelope the profile given by the first law; in fact $n = 1.35$ value should approximate this profile. But for different valve parameters, some discrepancies between the first law and polytropic models are noted.

CONCLUDING REMARKS

Based on our progress, the following tentative conclusions could be drawn.

1. First law and polytropic model match for p_c and m_c but not for T_c.

2. Cylinder temperature (T_c) prediction depends strongly on the mass flow rates ($\dot{m}_s$ and $\dot{m}_d$), valve deflections (q_s and q_d) and the heat transfer rate ($\dot{Q}$). Therefore, it is difficult to identify the "correct" temperature-time history.

3. Available empirical correlations for h_c are inadequate.

4. Until significant improvement is made in predicting cylinder temperature, there is little liklihood that detailed second law analyses can be performed.

Computational efforts to improve the cylinder temperature prediction, in tandem with an experimental program, are currently in progress.

LIST OF SYMBOLS

A Heat transfer area

c Specific heat

h Heat transfer coefficient

m Mass

n Polytropic index

p Pressure

q Valve displacement

Q Heat energy

R Gas constant

t Time

T Temperature

U Overall heat transfer coefficient

v Specific volume

W Work

z Compressibility factor

θ Crank angle

ω Running speed (rad/s)

Subscripts

a ambient

c cylinder

d discharge

o equilibrium (at p_s and T_s)

p pressure

s suction

v volume

Superscript

$\cdot$ d/dt

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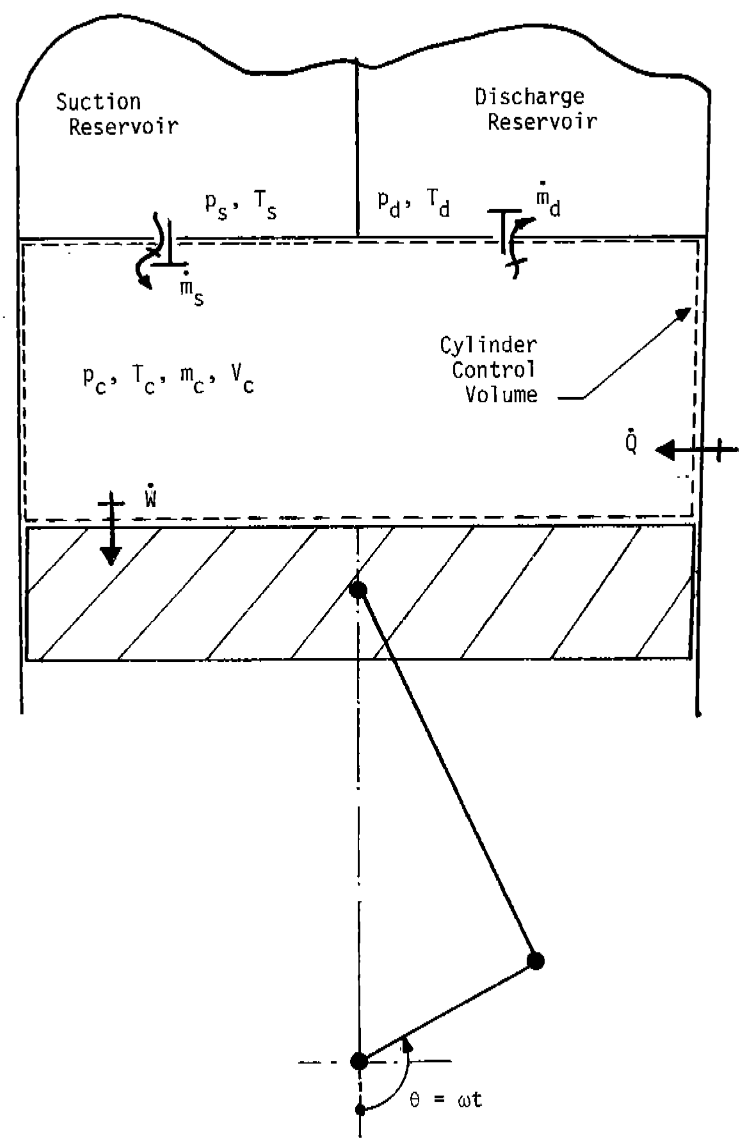

Figure 1. Cylinder control volume and basic variables

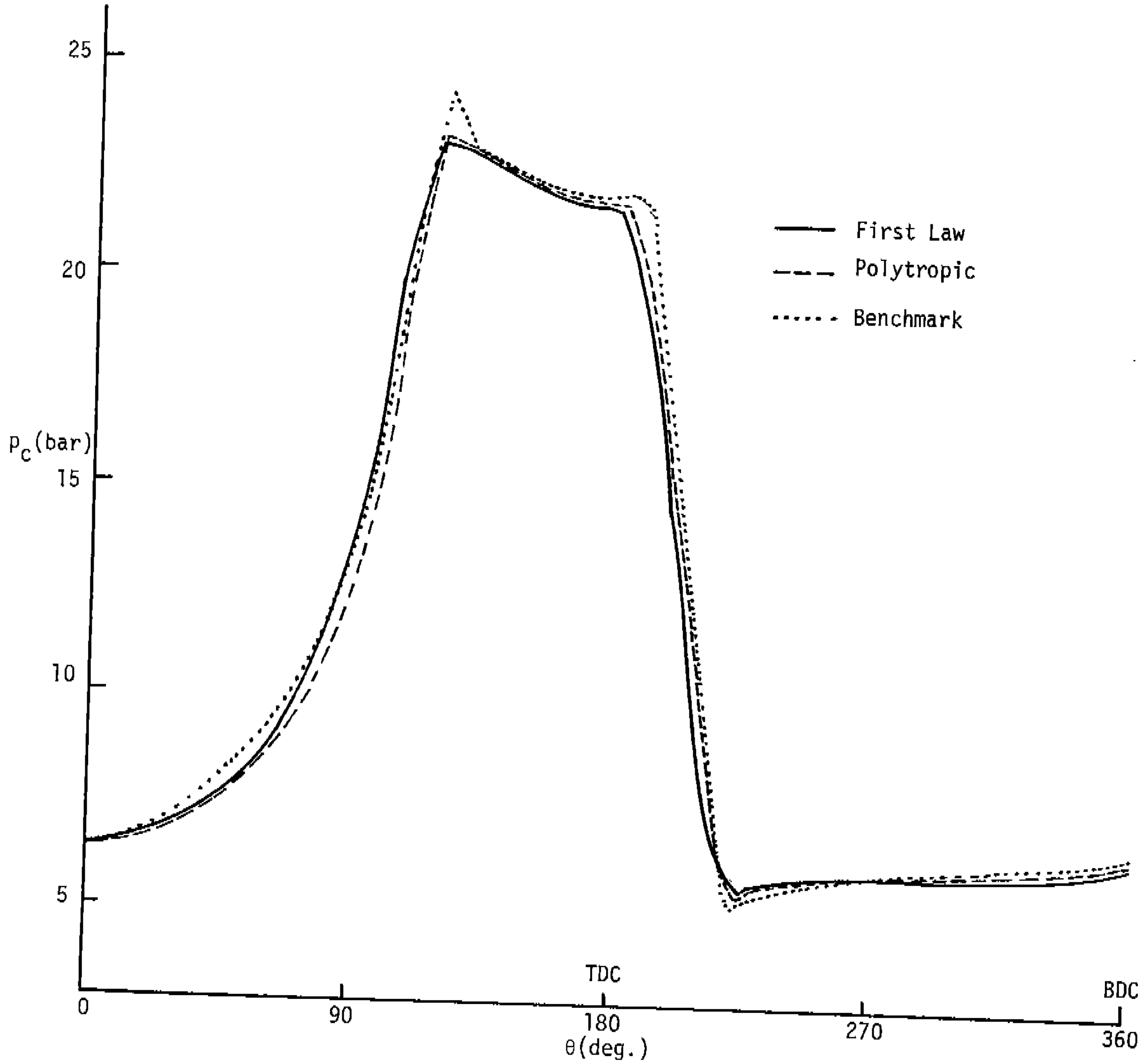

Figure 2(a). Comparison of single cylinder first law and polytropic models with benchmark simulation [20]. $P_c(\theta)$. Medium: R-22.

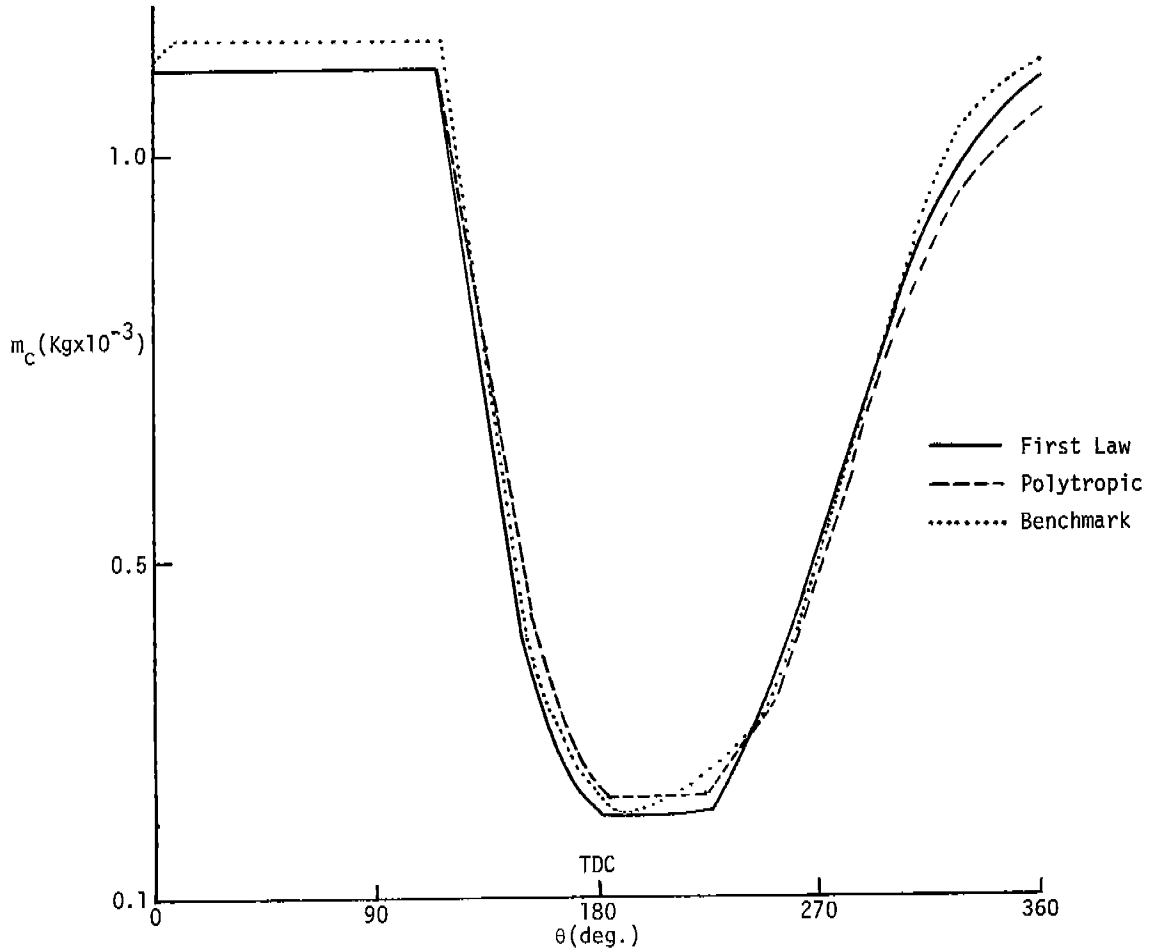

Figure 2(b). Comparison of single cylinder first law and polytropic models with benchmark simulation [20]. $m_c(\theta)$. Medium: R-22.

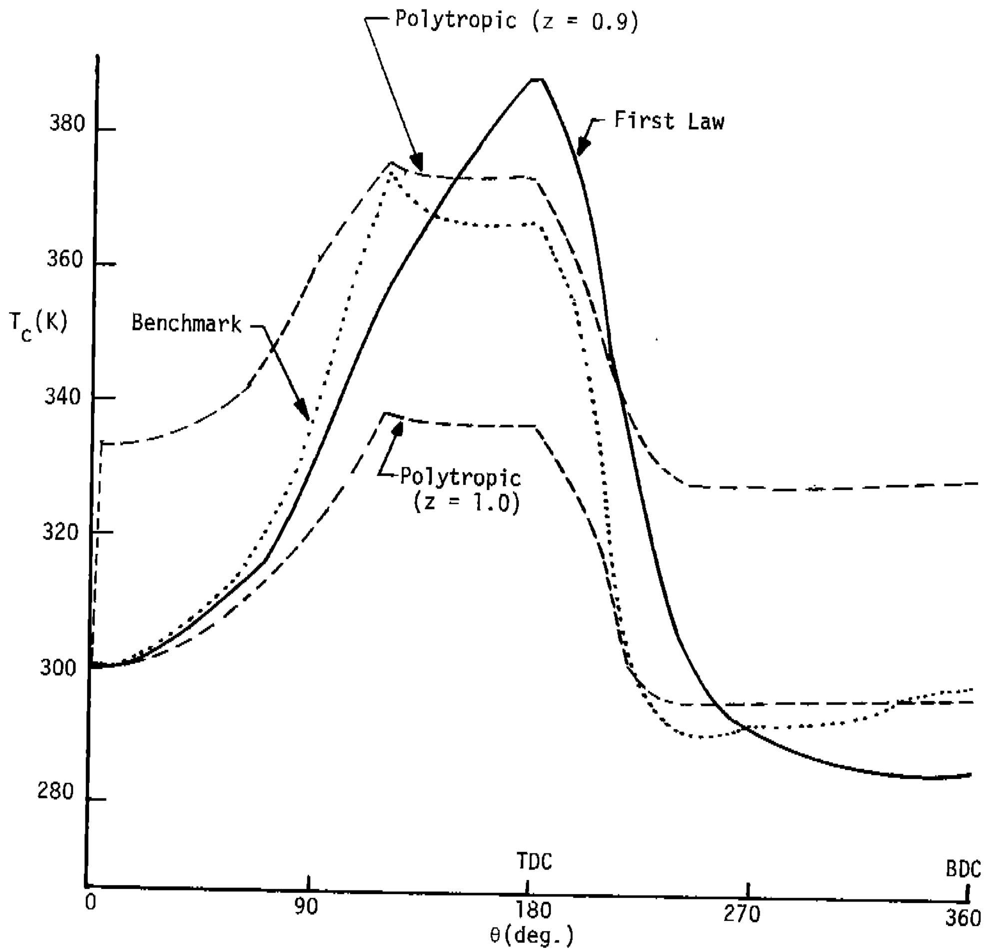

Figure 2(c). Comparison of single cylinder first law and polytropic models with benchmark simulation [20]. $T_c(\theta)$. Medium: R-22.

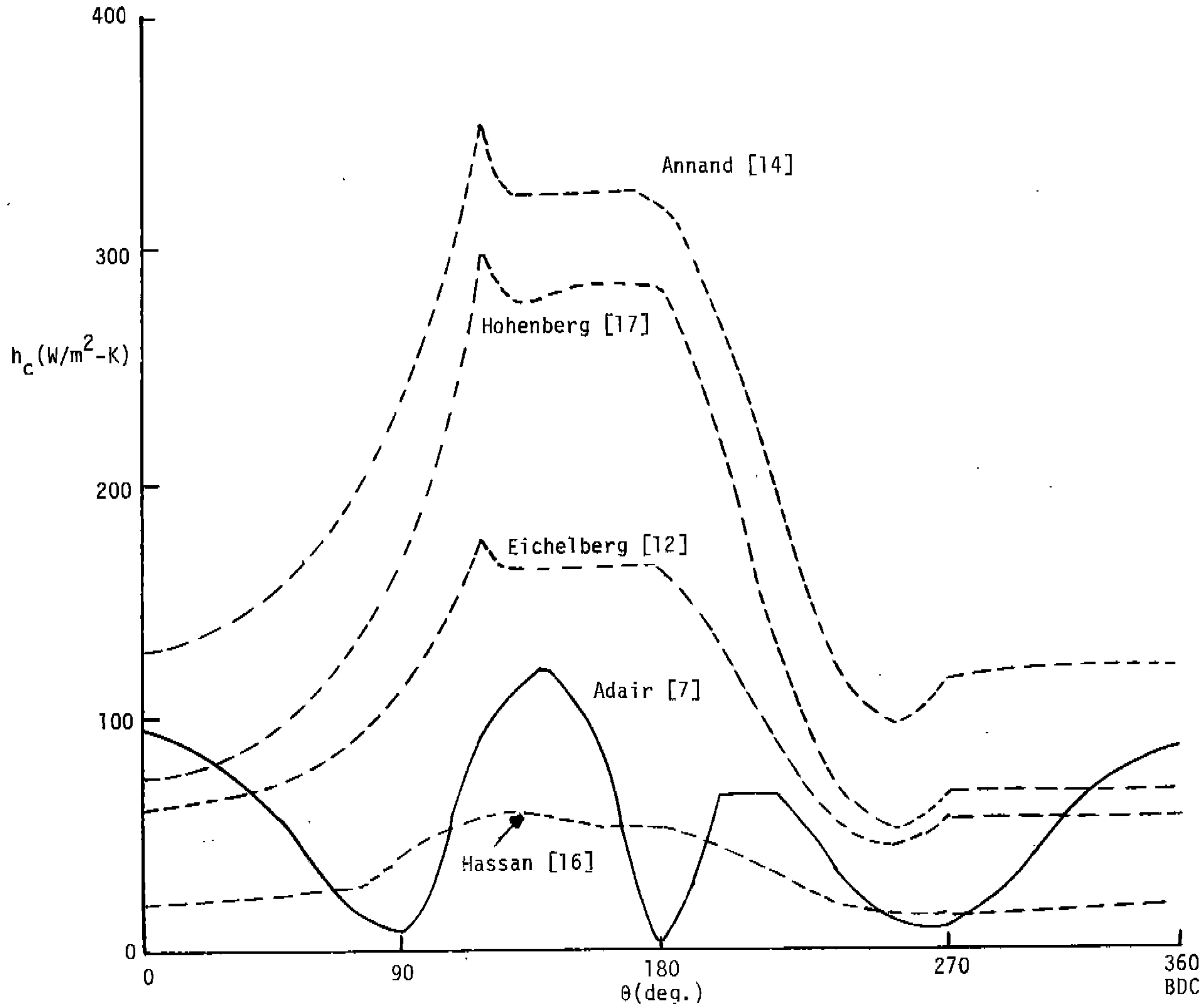

Figure 3(a). A comparison of cylinder heat transfer correlations. $h_c(\theta)$. Medium: Air.

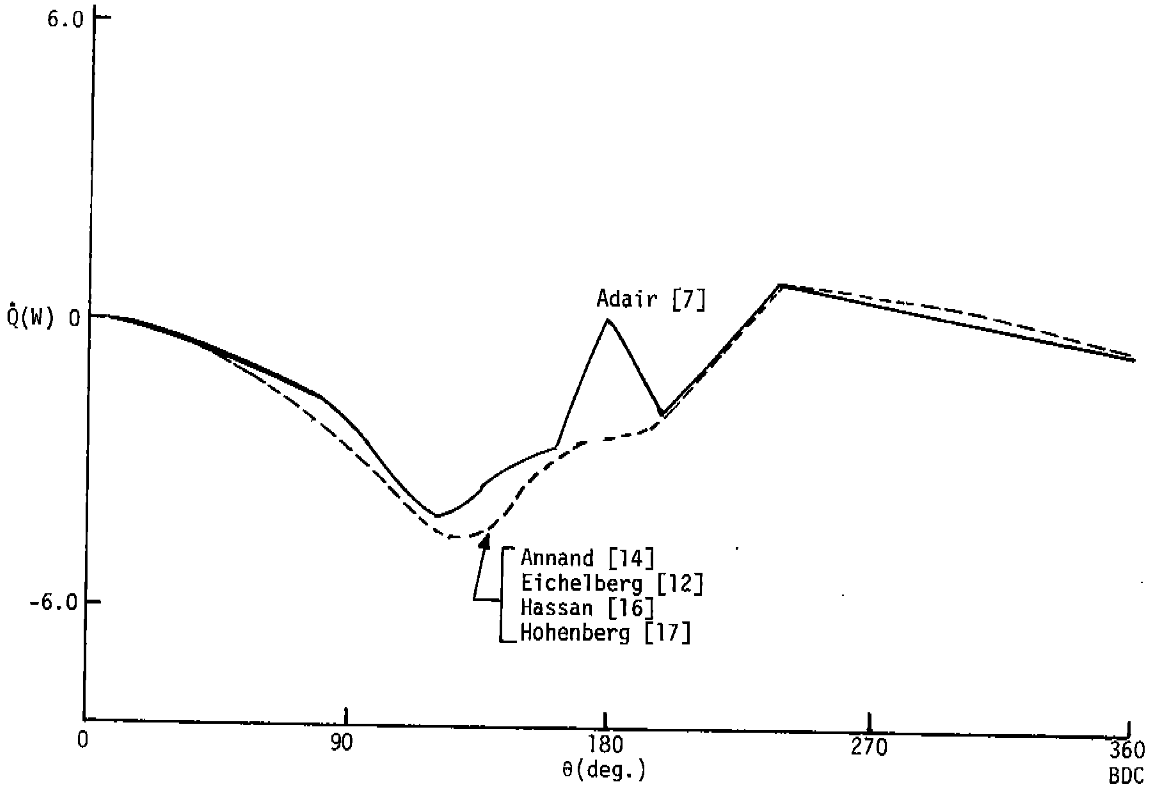

Figure 3(b). A comparison of cylinder heat transfer correlations. $\dot{Q}(\theta)$. Medium: Air.

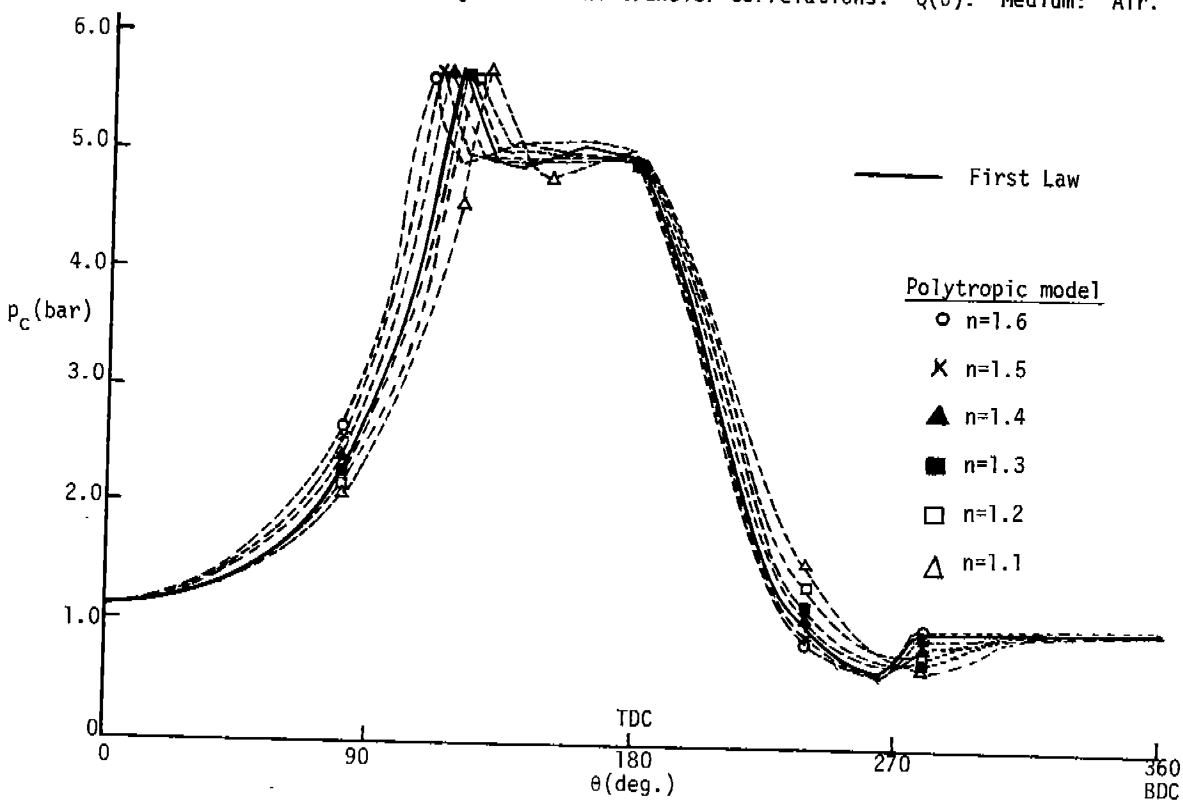

Figure 4(a). First law analysis vs. polytropic models. $p_c(\theta)$. Medium: Air.

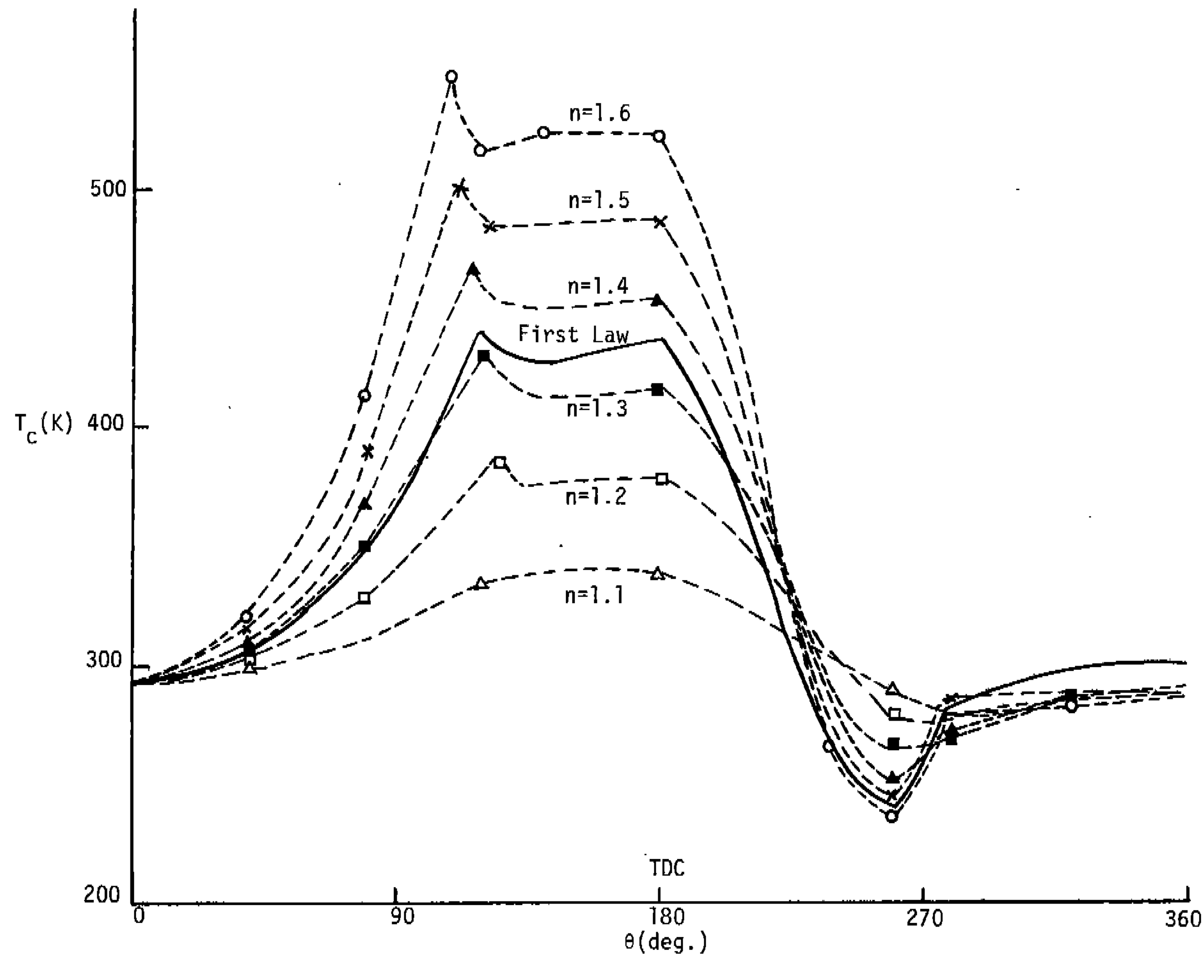

Figure 4(b). First law analysis vs. polytropic models. $T_c(\theta)$. Medium: Air.